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A Novel Model Considered Mass and Energy Conservation for Both Liquid and Vapor in Adsorption Refrigeration Systems

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ABSTRACT

In this paper, we proposed a dynamic model for a two-bed adsorption refrigeration system. Different from most existing researches which assume saturation vapor pressure in each device, the proposed method models the pressure in each device by considering both the liquid and vapor content in the device. Therefore, it can be more accurate in describing the system response and more suitable for studying the system instrumentation. The components included in this system model are: adsorption bed, evaporator, condenser, expansion valve, and etc. Each device is modeled based on the energy and mass conservation. Furthermore, the adsorption phenomenon is modeled by the "Freundlich equation," and "linear driving force model." The phase change of the refrigerant in evaporator and condenser is modeled by Hertz-Knudsen theory. In a case study, the pressure of the adsorption bed during the adsorption process is estimated to be 0.7kPa by the proposed model, while it was 1.6kPa by conventional method which assuming saturated vapor pressure. The coefficient-of-performance of the adsorption system is estimated to be 0.246 by this model, 0.36 by conventional method, and 0.28 by experimental data. The proposed model can estimate system performance more accurate than the conventional method. Moreover, the proposed model also inspire a new instrumentation strategy for the adsorption system, in which the system efficiency is improved and the pressure surge is avoided.

1. INTRODUCTION

Adsorption refrigeration system consists of sorption beds, evaporator, condenser, expansion valve, and etc. It can provide an effective way to transform waste heat, such as solar system, internal combustion engine (ICE), solid oxide fuel cell (SOFC), etc., into useful cooling power. Therefore, comparing to conventional refrigeration systems, the adsorption refrigeration system has a unique advantage in eco-friendly. Current research activities of the adsorption refrigeration system focus on two aspects. The first one is to decrease the operating temperature of the adsorption bed, in which the lower grade of waste heat can be used to drive the system. For example, by using different adsorption material or multi stage adsorption techniques (Saha *et al.*, 1995a), the operation temperature can be reduced to about 50℃, which is close to the temperature of a solar water heater system. The second one is to improve the coefficient of performance (COP) of the refrigeration system. This factor is always important for discriminating a refrigerator is good or bad, and its value of an adsorption refrigeration system is around 0.5~0.6. To improve the COP of a system, several system architectures and operation techniques have been proposed, such as multi-bed adsorption, heat recovery cycle (Wang and Chua, 2007), thermal wave techniques (Sward et al., 2000), cascading adsorption system (Liu and Leong, 2006), and etc. Since an adsorption system consists of many components, no matter in which research direction, a precise mathematic model of the overall system always plays an important in the system development.

Among most of the modeling work of the adsorption refrigeration system presented in paper, a common assumption is that the pressure in each component is determined either by the saturation vapor pressure of the condenser or evaporator. Under that assumption, the work of system modeling can be quite simple because the pressure of the adsorption bed is either the same as the condenser or evaporator, and the mass conservation is considered only in calculating the refrigerants in "adsorbed phase" and "liquid phase" (Liu and Leong, 2006). But, the pressure of the sorption bed would not be the same as the pressure of its upstream (evaporator) or downstream (condenser) devices. Therefore, this error would lead to errors in estimating the amount of refrigerant cycling in the system. To improve the modeling accuracy, the mass conservation and energy conservation are considered in modeling each system component. And, the flow rate between each component is modeled by "Bernoulli's equation." Furthermore, the adsorption/ desorption process is modeled by "Hertz-Knudson theory" (Badam et al., 2007).

2. SYSTEM DESCRIPTION

2.1 Working principle

As shown in Figure 1, the adsorption refrigeration system considered in this paper includes two sorption beds, evaporator, condenser, expansion valve, and four control valves. The complete adsorption refrigeration cycle requires four thermodynamic processes which are adsorption, heating, desorption, and cooling. During the adsorption process, the valve 1 is open and the valve 2 is closed. The refrigerant, which is water in this case, evaporate in the evaporator and adsorbed by silica gel in the adsorption bed 1. During the heating process, the valve 1 and 2 are both closed. The adsorption bed 1 is then heated up by heat exchanger and the temperature and pressure of the adsorption bed is raised. During the desorption process, the valve 2 is open and valve 1 is closed. The water vapors start to desorb from the adsorbent and flow to the condenser. Water vapors condense to liquid and then flow back to evaporator through expansion valve. The adsorption bed 2, valve 3, and valve 4 works in an alternating manner such that the adsorption bed 2 is in the adsorption process when the adsorption bed 1 is in the desorption process.

2.2 Assumptions for system modeling

Several assumptions are made when developing a mathematic model for the refrigeration system. (1) Temperature and pressure in each device are homogeneous. (2) The water vapor in the piping system is ignored. (3) Temperature of the liquid and vapor are the same for each device. (4) No heat loss from the evaporator, condenser, and adsorption beds. (5) Properties such as specific heat capacity, heat transfer coefficient, etc. are considered to be constant.

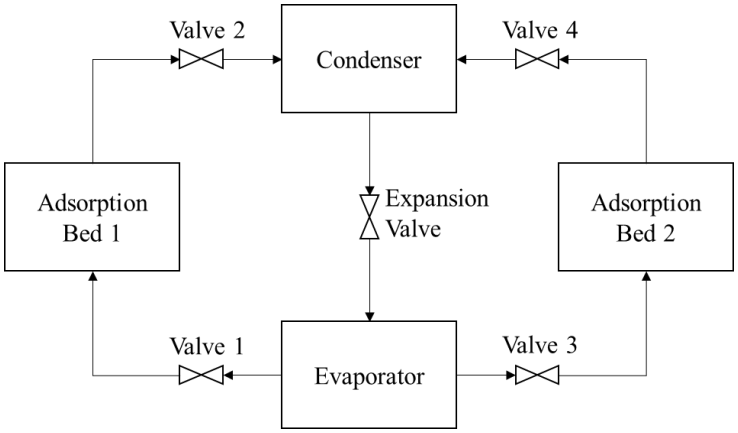

Figure 1: A schematic of a two-bed adsorption refrigeration system.

3. MATHEMATICAL MODELING

3.1 Mass flow rate between components

In this paper, we use the Bernoulli's equation to model the flow rate of the water between each system component. If the viscous loss, height difference of the flow are ignored, and the flow rate in downstream is assumed to be much larger than in upstream, the mass flow rate can be modeled as follows.

$$\dot{m}_{downstream} = A\sqrt{2\rho\left(P_{up} - P_{down}\right)} \tag{1}$$

where $\dot{m}_{downstream}$ is mass flow rate at downstream of the piping system, *A* is the cross sectional area of the pipe, ρ is density of the flow, and P_{up} & P_{down} are upstream and downstream pressure, respectively.

3.2 Adsorption bed

To model the adsorption phenomenon, we use the Freundlich equation (Chihara and Suzuki, 1983) to describe the amount of adsorption when the system is at equilibrium.

$$x_s = x_0\left(\frac{p_b}{p_a}\right)^{1/n} \tag{2}$$

where x_s is the amount adsorption per kilogram of adsorbent in the saturation condition, x_0 is the maximum amount of adsorption per kilogram of adsorbent, P_a is the saturation pressure of water at the temperature of the adsorbent, P_b is the pressure of the adsorption bed, and $1/n$ is material constant. The values of x_0 and $1/n$ are functions of the adsorbent temperature, which can be represented in Equation 3 and 4 (Saha et al., 1995b).

$$x_0(T_a) = A_0 + A_1T_a + A_2T_a^2 + A_3T_a^3 \tag{3}$$

$$\frac{1}{n} = B_0 + B_1T_a + B_2T_a^2 + B_3T_a^3 \tag{4}$$

where T_a is the temperature of the adsorbent, $A_0 \sim A_3$ and $B_0 \sim B_3$ are the coefficient to be determined by experiment (Saha et al., 1995b). The values of these coefficients used in this paper are listed in Table 1. The frequently used "linear driving force (LDF)" method is used to describe the transient response of the adsorption (Wang and Chua,2007).

$$\frac{dx}{dt} = K(x_s - x) \tag{5}$$

$$K = \frac{15 \times D_{S0} \times \exp\left(-\frac{E_a}{RT_a}\right)}{R_p^2} \tag{6}$$

where x is the amount adsorption per kilogram of adsorbent, K is the mass transfer coefficient, D_{S0} is the surface diffusion coefficient, E_a is surface affinity, R is the gas constant of the water, R_p is the average diameter of pores in the adsorbent.

Other than the adsorption behaviors, the mathematics modeling of the adsorption bed includes the continuity equation, the first law of thermodynamic, and the phase change between liquid and vapor. For the mass balance equation:

$$\frac{dW_a}{dt} = \dot{m}_1 - \dot{m}_2 - M_b\frac{dx}{dt} \tag{7}$$

where W_a is mass of water vapor in the adsorption bed, $\dot{m}_1$ is mass flow rate from the evaporator to adsorption bed, $\dot{m}_2$ is the mass flow rate from the adsorption bed to condenser, M_b is the mass of the adsorbent in the adsorption bed. For the energy balance equation:

$$\left(M_{Cu}C_{Cu} + M_{Al}C_{Al} + M_bC_b + M_bxC_{pg}\right)\frac{dT_a}{dt} = \dot{m}_1h_g(T_e) - \dot{m}_2h_g(T_a) - M_b\frac{dx}{dt}h_{ads} + E_bC_{pf}\dot{m}_w(T_{wh} - T_a) \tag{8}$$

where M_{Cu} and M_{Al} are mass of copper tube and aluminum fin of the heat exchanger, C_{Cu} and C_{Al} are the respective specific heat capacity of copper and aluminum, C_b is the specific heat capacity of the adsorbent, C_{pf} and C_{pg} are respective specific heat capacity of water liquid and vapor, T_{wh} is the inlet temperature of the heat exchanger, E_b is the effectiveness of heat exchanger that can be calculated using equation (9), h_{ads} is adsorption heat for phase change (Sakoda and Suzuki, 1984), $\dot{m}_w$ is the flow rate of the heat exchanger.

Table 1: Characteristic coefficient for adsorbent-adsorbate material (B.B. Saha et al., 1995b)

$A_0 = -6.5314$	$B_0 = -15.587$
$A_1 = 8.2452 \times 10^{-2}$	$B_1 = 1.5915 \times 10^{-1}$
$A_2 = -2.3951 \times 10^{-4}$	$B_2 = -5.0612 \times 10^{-4}$
$A_3 = 2.5493 \times 10^{-7}$	$B_3 = 5.3290 \times 10^{-7}$

$$E_b = 1 - \exp\left(-\frac{UA}{\dot{m}_w C_{pf}}\right) \tag{9}$$

where *U* is the overall heat transfer coefficient. *A* is the area for the heat exchange.

3.3 Evaporator

We use a simplified Hertz-Knudsen model to simulate the rate of evaporation. After assuming the evaporation coefficient equals to one, the amount of evaporation can be described as follows.

$$\dot{m}_p = A_e \sqrt{\frac{M}{2\pi R T_e}} \left[P_{sat}(T_e) - P_v\right] \tag{10}$$

where $\dot{m}_p$ is phase changing rate, *M* is molecular weight of water, T_e is the temperature of water in the evaporator, $P_{sat}(T_e)$ is the saturation vapor pressure at the temperature T_e, P_v is vapor pressure of the evaporator. The total mass of refrigerants in the evaporator is divided into liquid and vapor. Using Equation 11~13, one can calculate the change rate of the refrigerant in the evaporator.

$$\frac{dW_e}{dt} = \frac{dW_{ef}}{dt} + \frac{dW_{eg}}{dt} \tag{11}$$

$$\frac{dW_{ef}}{dt} = \dot{m}_{exp} - \dot{m}_p \tag{12}$$

$$\frac{dW_{eg}}{dt} = \dot{m}_p - \dot{m}_1 \tag{13}$$

where W_e is the total mass of refrigerant in evaporator, W_{ef} and W_{eg} are the respective mass of liquid and vapor, $\dot{m}_{exp}$ is the mass flow rate from the expansion valve. For the energy conservation, one can write down the following equations.

$$\left(M_{Cu} C_{Cu} + W_e C_{pw}\right) \frac{dT_e}{dt} =$$
$$\dot{m}_{exp} h_f(T_c) - \dot{m}_1 h_g(T_e) + E_e C_{pf} \dot{m}_e (T_{ei} - T_e) - \dot{m}_p h_{fg}(T_e) \tag{14}$$

where E_e is the effectiveness of the heat exchanger, $\dot{m}_e$ is the mass flow rate of the heat exchanger in the evaporator subsystem, h_{fg} is the latent heat of the water.

3.4 Condenser

The working principle of the condenser is almost the same as that of the evaporator. The major difference between these two devices is that the phase change and heat exchange take place in the opposite direction. Therefore, the mathematic model of the condenser is almost identical to that of the evaporator, with the subscription of the system parameters changing from "*e*" to "*c*". The model of the condenser can be written as follows.

$$\dot{m}_p = A_c \sqrt{\frac{M}{2\pi R T_c}} \left[P_v - P_{sat}(T_c)\right] \tag{15}$$

$$\frac{dW_c}{dt} = \frac{dW_{cf}}{dt} + \frac{dW_{cg}}{dt} \tag{16}$$

$$\frac{dW_{cf}}{dt} = -\dot{m}_p - \dot{m}_{exp} \tag{17}$$

$$\frac{dW_{cg}}{dt} = \dot{m}_2 + \dot{m}_p \tag{18}$$

$$\left(M_{Cu} C_{Cu} + W_c C_{pw}\right) \frac{dT_c}{dt} =$$
$$\dot{m}_2 h_g(T_b) - \dot{m}_{exp} h_f(T_c) - E_c C_{pw} \dot{m}_c (T_{ci} - T_c) + \dot{m}_p h_{fg}(T_c) \tag{19}$$

Table 2: Physical property values

Values used in simulation					
Symbol	**Unit**	**Value**	**Symbol**	**Unit**	**Value**
A_c	m2	3.73	U_{des}	kW/m2•K	1.72414
A_e	m2	1.91	U_{eva}	kW/m2•K	2.55754
A_a	m2	2.46	h_{ads}	kJ/kg	$2.8^{\times}103$
A_{exp}	m2	$3.85^{\times}10\text{-}5$	Rp	m	$7.1^{\times}10\text{-}4$
$C_{p\,Al}$	kJ/kg•K	0.905	$M_{con\,Cu}$	kg	24.28
$C_{p\,Cu}$	kJ/kg•K	0.386	$M_{eva\,Cu}$	kg	12.45
C_{wf}	kJ/kg•K	4.18	W_{ef}	kg	50
C_{wg}	kJ/kg•K	1.85	Ma_{Al}	kg	64.04
C_{pb}	kJ/kg•K	0.924	Ma_{Cu}	kg	51.20
D_{so}	m2/s	$2.54^{\times}10\text{-}4$	M_b	kg	47
Ea	J/mol	$4.2^{\times}104$	Va	m3	$6.046^{\times}10\text{-}3$
U_a	kW/m2•K	1.60256	Vc	m3	$1.622^{\times}10\text{-}2$
U_{con}	kW/m2•K	4.11523	Ve	m3	$8.318^{\times}10\text{-}3$
$\dot{m}_w$	m/s	1.3	adsorption/desorption time	s	420
$\dot{m}_c$	m/s	1.6	heating/cooling time	s	30
$\dot{m}_e$	m/s	0.7			

3.5 Expansion valve

The flow rate of the throttling valve is similar to Equation 1, but needs to add a constant to simulate the intended energy loss in an expansion valve design.

$$\dot{m}_{exp} = CA\sqrt{\rho(P_c - P_e)} \tag{20}$$

where *C* is the discharge ratio of the expansion valve.

4. RESULT AND DISCUSSION

In order to track the effect due to complicated system modeling, the values of system parameters used in this model are obtained from the work of B.B. Saha et al. (1995a & b) and Wang and Chua (2007). These values are listed in Table 2 for reference. The following properties are observed to justify the feasibility of the proposed model which includes the pressure of each system component, and COP, cooling power under different operation conditions.

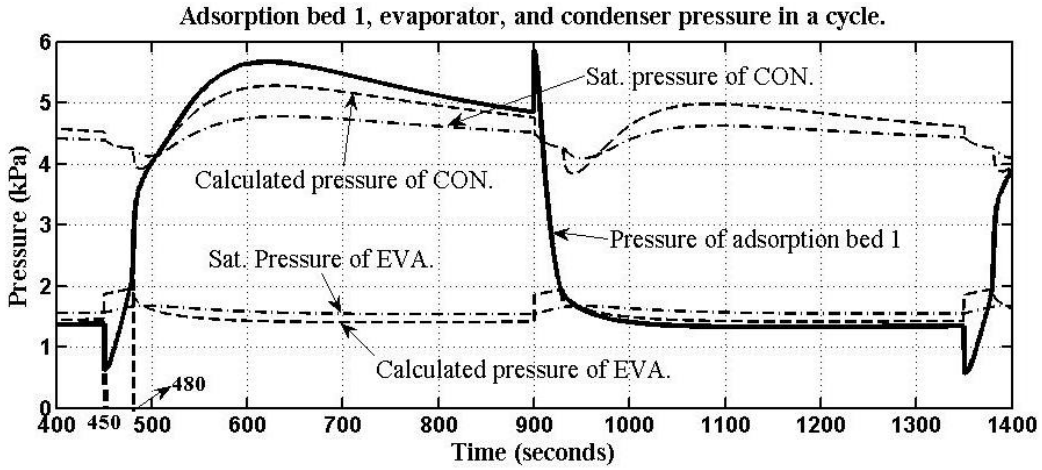

Figure 2: The pressure of the adsorption bed, evaporator, and condenser when operating the system by conventional approaches.

4.1 Pressure in the each system component

Figure 3 shows the pressure of the adsorption bed 1, condenser and evaporator within a complete thermal cycle. The adsorption bed 1 is at the adsorption process before the 450 second. The valve 1 closes at the 450 second and the system enters the heating process. After 30 seconds of the heating process, the valve 2 opens and the system enters the desorption process. At the 900 second, valve 2 is closed and the system is ready for the next adsorption process.

According to the simulation results, this model predicts the operation pressure of the condenser that is larger than its saturation pressure. And, the operation pressure of the evaporator is slightly smaller than its saturation pressure. Therefore, the amount of refrigerant cycling in this system, predicted by this model, would be less than the amount predicted by the conventional models which use the saturation pressures for the condenser and evaporator.
Furthermore, at the 480 second when the valve 2 is open, the pressure of the adsorption bed is less than that of the condenser. Therefore, the refrigerant may flow in the reverse direction for a couple seconds which may decrease the system efficiency. Besides, since the valves do not operate at the proper timing, the pressure surge happens at the 450 second and 900 second, which may damage the adsorption bed. These simulation results illuminate the importance of the accurate pressure modeling of the adsorption bed.

Figure 4 shows a new instrumentation strategy for operating the adsorption system, which is suggested by this system modeling work. In this new approach, the system still enters the heating process at the 450 second. And, the valve 1 closes at 475.2 second when the pressure of the adsorption bed is about to be higher than the pressure of the evaporator, the valve 2 opens at 508.3 second when the pressure of the adsorption bed is about to be higher than that of the condenser. In this case, the refrigerant would flow in the designated direction. The system efficiency can be improved and the pressure surge is avoided.

4.2 COP values under different operation conditions

In this case study, the temperature of the hot water in the heating process is set to be either 85℃ or 70℃, and cooling water in the cooling process is set to be either 32℃ or 30℃. The cold side of the heat exchange process of the evaporator ranges from 10℃ to 16℃. The predicted COP and cooling power under different temperature conditions are shown in Figure 5 and 6. The legend "COP_e" are the experimental results obtained by Boelman et al. (1995), "COP_ref" are the COP value predicted by the conventional model (Boelman et al. ,1995), and "COP_s" are the COP value predicted by this model. According to the simulation results shown in Figure 5 and 6, the proposed method can estimate the COP value more close to the experimental results than the conventional model in most of the conditions. For example, in the case of Figure 6 (a), the COP of the adsorption system is estimated to be 0.246 by this model, 0.36 by conventional method, and 0.28 by experimental data. However, the cooling power predicted by this model is not as accurate as predicted by the conventional model. This model tends to underestimate the cooling power as compared to the experimental data. This deficiency is still under investigation.

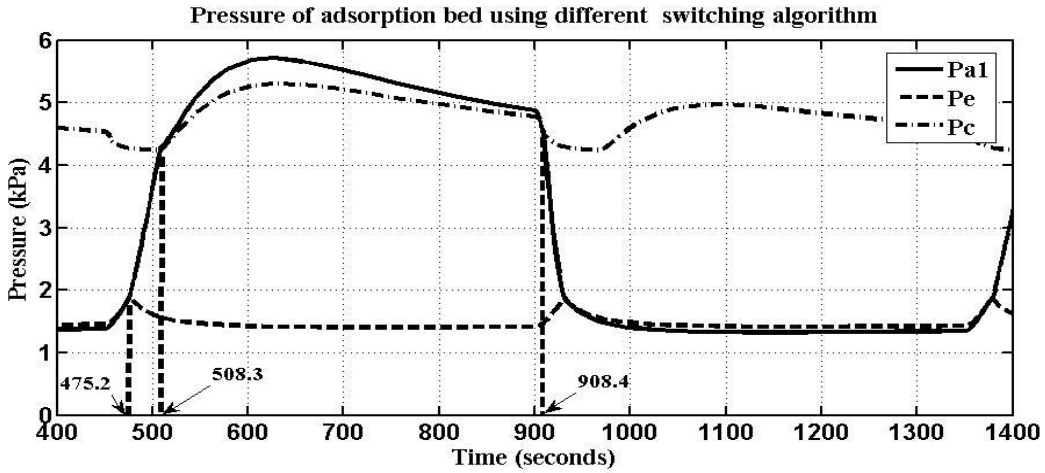

Figure 3: The pressure of the adsorption bed, evaporator, and condenser when operating the system by new instrumentation approaches.

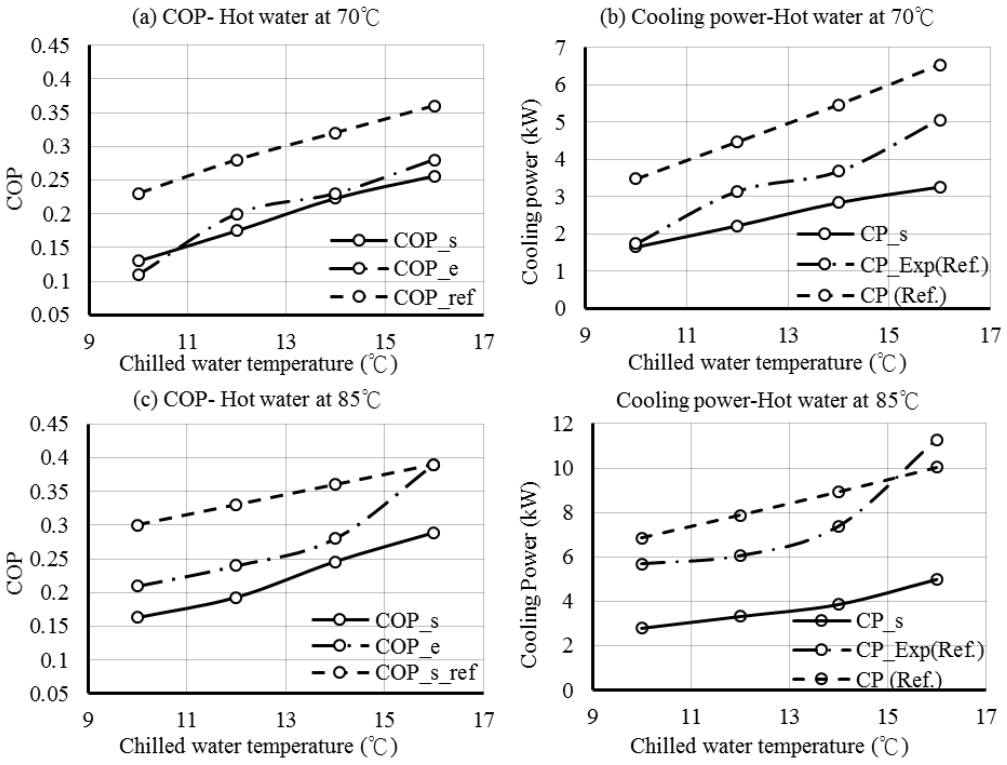

Figure 4: COP and cooling power of the system under different operating temperatures. Cooling water in the cooling process are at **32℃**. The hot water in the heat process are at 70℃ in (a)(b) and 85℃ in (c)(d).

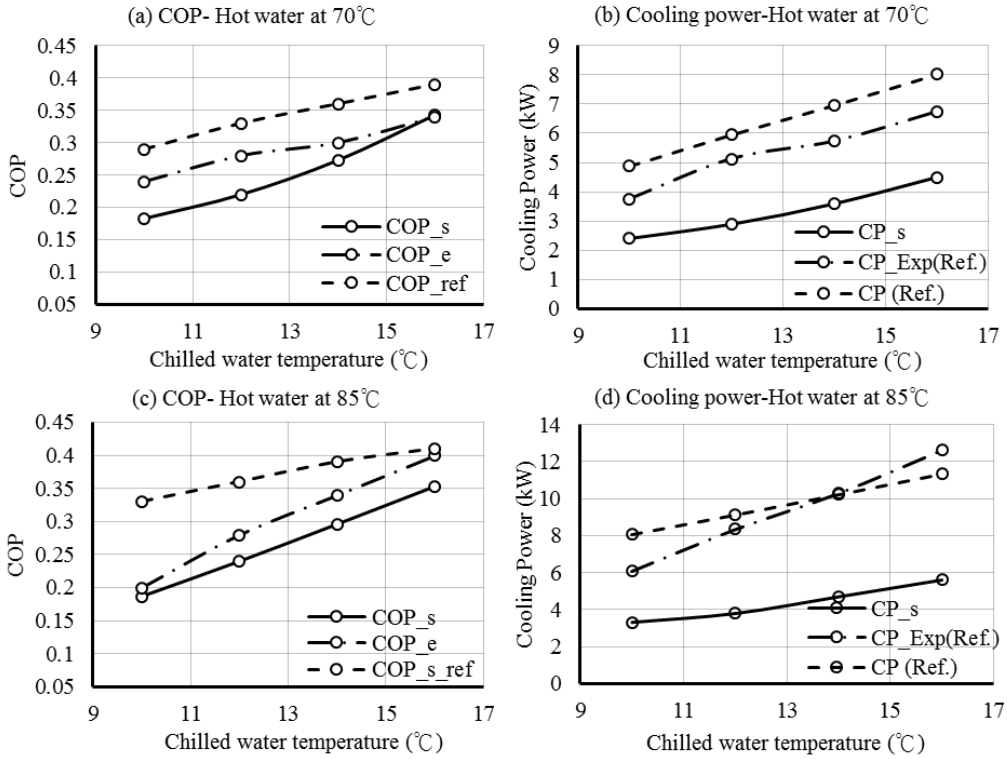

Figure 5: COP and cooling power of the system under different operating temperatures. The cooling water in the cooling process are at **30℃**. The hot water in the heating process are at 70℃ in (a)(b) and 85℃ in (c)(d).

5. CONCLUSIONS

This paper proposed a dynamic model for a two-bed adsorption refrigeration system, in which the mass conservation and energy conservation are used to model the liquid and vapor content in each device. This new system model suggested that the conventional mathematic model may overestimate the amount of the refrigerant cycling in the refrigeration system, which results in the overestimation of the COP and cooling power of the system. In a case study, the COP of the adsorption system is estimated to be 0.246 by this model, 0.36 by conventional method, and 0.28 by experimental data.

The proposed model also suggested that the adsorption refrigeration system may experience pressure surge and low efficiency if the pressure of the adsorption bed is not modeled accurately and the control valves do not operated at the correct timing. A modified instrumentation strategy is proposed in which the operation of the control valves is determined by the pressure of the adsorption bed, instead of the onset of the heating and cooling processes.

NOMENCLATURE

A	heat transfer area	(m^2)
C	discharge coefficient	(dimensionless)
C_P	specific heat capacity	(KJ/Kg-K)
CON.	Condenser	
D_{S0}	surface diffusion coefficient	(m^2/s)
E	effectiveness	(dimensionless)
E_a	surface affinity	(J/mol)
EVA.	Evaporator	
K	mass transfer coefficient	(1/s)
M	mass of solid	(kg)
m	mass flow rate	(kg/s)
P	pressure	(kPa)
R	gas constant	(kJ/kg-K)
R_P	adsorbent particle diameter	(m)
Sat.	Saturation	
T	temperature	(K)
U	heat transfer coefficient	(W/m^2K)
W	mass of fluid	(kg)
h	adsorption heat	(kJ/kg-K)
x	amount adsorbed	(kg/kg)
ρ	fluid density	(kg/m^3)

Subscript

a	adsorption bed
ads	adsorption
Al	aluminum
b	adsorbent
c	condenser
Cu	copper
e	evaporator
exp	expansion valve
f	fluid
g	gas
i	inlet
sat	saturation
w	water
1~4	numbers of valves

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